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This material was prepared for use in a lecture at the Oak Ridge School of Reactor Technology and at the University of Tennessee Extension Division course in nuclear power plant design.

I. BASIC PRINCIPLES HEAT POWER CYCLES

In designing the reactor for an atomic power plant the nuclear scientist or engineer must consider the influence of his design on the performance of the associated heat power cycle. Substitution of the nuclear furnace for the combustion process, while simple in principle, changes cost patterns, response to load changes, and realizable thermal efficiencies significantly, so that precepts once thought to be well established in power plant design may require reappraisal. Thermodynamic study of the various heat power cycles lends an understanding of the direction one should take to improve the plant over-all efficiency, but as to be expected, only through an appreciation of the engineering and economics of central power stations can such studies be meaningful. The financial aspects (see Section IX) need particular emphasis, for the large outlays required for modern heat-power stations make the stakes quite high, where only a fraction of a percent difference in overall plant efficiency or in an estimated yearly load factor, can represent hundreds of thousands of dollars annually. In the following discussion of elementary basic principles of heat power cycles with regard to relative thermal efficiency, one must not lose sight of the fact that the immediate and pressing problem is to build workable reactors of demonstrated reliability for central power station operation - that small differences in predicted thermal efficiency are relatively unimportant at this stage in the development.

On the basis of present knowledge, direct conversion of nuclear fission energies into electrical energy, while extremely attractive, appears to hold little promise. The nuclear fusion process is in even the more

distant future. It must be accepted, then, that the primary output of a nuclear reactor is in the form of heat. Direct conversion of heat into electrical energy through thermoelectric processes has been given considerable study, but because of the low thermal efficiencies (3 to 7%) this method apparently has little application. Practical thermal efficiencies are presently available only through conversion of the heat into mechanical work as an intermediate step in the production of electrical power.

The methods of converting heat into useful work may be put into two broad categories, (1) those in which the working fluid exists in a single-phase, such as in the gas turbine cycle, and (2), those in which the working fluid undergoes phase change, such as in the steam cycle. It will be well to first briefly review some of the very elementary thermodynamic principles without regard for the working fluid used.

The Carnot Cycle shown in Fig. 3 is the most efficient heat-power cycle that can be devised. This cycle consists of adding heat to the working fluid at constant temperature, T_1 , expansion of the fluid in an adiabatic and frictionless engine, rejection of the heat to a heat sink at constant temperature, T_2 , followed by compression of the working fluid to its original state point in an adiabatic and frictionless compressor, or pump. When the temperatures of the heat source and heat sink are expressed as absolute temperatures, the thermal efficiency of the Carnot cycle is:

$$\eta = \frac{T_1 - T_2}{T_1}$$

This Carnot efficiency is one that can be approached but not exceeded. Typical values for the heat sink (ambient air, water, etc.) are 65 to 70°F, so that for a heat source of 500°F the maximum attainable efficiency is 56%;

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Fig. 1. The Carnot cycle.

Fig. 2. Turbine and compression efficiencies.

for a 1000°F source, is 64%. The actual efficiency of a plant divided by the calculated Carnot efficiency is often termed the percent of perfection, and it is interesting to note that within the past thirty years steam power plants have progressed from about 34% perfection to slightly over 60% (see Table 1).

Entropy, "s", is defined as the ratio of the quantity of heat added to or lost from a pound of the working fluid to the absolute temperature at which the heat flows, that is:

$$ds = dQ/T = \text{Btu/lb-}^{\circ}\text{F}_{\text{abs}}$$

When the entropy is plotted against the absolute temperature, T , as in Figure 1, the area represents heat. The area under the line 1-2 is heat added to the cycle, and the area under line 3-4 is the heat rejected by the cycle; the enclosed area 1-2-3-4 is the useful work of the cycle, expressed in heat units. An expansion or compression process that is adiabatic and reversible (i.e., frictionless, etc.) is one of constant entropy. Since this perfect process does not exist in nature, the entropy always increases in actual expansion or compression, as indicated in Figure 2.

Enthalpy, "h", represents the heat energy at any given condition of temperature, pressure, etc., and in the case of a gas is made up of two parts: one is the internal energy which results from the molecular motions in the fluid and is a function only of the temperature; the other is the work, or p_v energy, representing the work put into the gas to cause it to reach any given volume when opposed by the external pressure under which it is contained. The enthalpy of steam contains three elements:

(1) the heat of saturated liquid, h_f , which is the heat required to raise the temperature of liquid water from the datum temperature (32°F) to the boiling temperature at the given pressure; (2) the heat required to evaporate the water completely at the given pressure, called the internal latent heat of the steam, which must be combined with the external work term from the energy done by the steam in making room for itself in its environmental pressure, the sum of these two energies being the total latent heat of vaporization, h_{fg} ; and (3), the energy required to raise the temperature of the completely evaporated water from that at boiling to some degree of superheat, which, as for any gas, is a function of the specific heat and the temperature change. The total enthalpy of saturated steam is $h_g = h_f + h_{fg}$. The total enthalpy of wet steam is: $h_x = h_f + x(h_{fg})$, where x is the fraction of the one pound of water that has been evaporated, which on a percentage basis is known as the quality of the steam. The total enthalpy of superheated steam is: $h = h_f + h_{fg} + h_{\text{superheat}}$. The heat added to water in a boiler may be found by taking the difference between the enthalpy entering and the total enthalpy of the steam leaving. The work delivered per pound of working fluid flowing through a turbine is the difference between the total enthalpies entering and leaving (if velocities are neglected). Plots of enthalpy vs entropy (h-s plane), or in the case of water, more commonly-called the Mollier diagram, have no significance in regard to areas, but are very useful in showing heat power cycle processes. (The thermodynamic properties of water are conveniently arranged in Thermodynamic Properties of Steam*, and the properties

* Keenan, Joseph E., and Keyes, F. G., Thermodynamic Properties of Steam, John Wiley & Sons, 1936.

of air and other gases are in Gas Tables*.)

Turbine efficiency is the ratio of the actual work done by a unit mass of any working fluid in expanding through a prime mover, to the amount that could have been done had the process been isentropic (adiabatic and reversible), see Fig. 2.

$$\eta_{\text{tur.}} = \frac{\Delta h_{\text{actual}}}{\Delta h_{ds=0}} = \frac{h_1 - h_2}{h_1 - h_2'}$$

The compression efficiency is the ratio of the isentropic work, $\Delta h_{ds=0}$, to the work actually required for a compression process, see Fig. 2.

$$\eta_{\text{comp.}} = \frac{\Delta h_{ds=0}}{\Delta h_{\text{actual}}} = \frac{h_3 - h_4'}{h_3 - h_4}$$

The thermal efficiency of a hypothetical vapor cycle, shown in Fig. 3 as 1-2'-3'-4', is called the Rankine Cycle efficiency. The cycle consists of, 1-2', isentropic expansion through a turbine; 2'-3', constant pressure condensation of the steam to saturated liquid; 3'-4', isentropic pumping of the liquid water to boiler pressure; 4'-1, constant pressure addition of heat to the initial conditions.

$$\eta_{\text{RC}} = \frac{h_1 - h_2 - (h_4' - h_3')}{h_1 - h_4'}$$

* Keenan, Joseph H., and Kaye, Joseph, Gas Tables, John Wiley & Sons, 1948.

The pump work is small and is sometimes neglected. The efficiency of the Rankine Cycle is less than the Carnot efficiency because the heat is not all added to the system at the maximum temperature, T_1 .

Example 1. -- Steam leaves a steam generator at 400 psia and 500°F and expands in a turbine to a pressure of 2" Hg abs and a moisture content of 12%. Water enters the steam generator at 90°F. Ignoring boiler feed-pump work, what is (a) the turbine efficiency, (b) the Rankine Cycle efficiency, (c) the actual cycle thermal efficiency?

(a) At initial conditions:

$$p_1 = 400 \text{ psia.}$$

$$h_1 = 1245.1 \text{ Btu/lb}$$

$$t_1 = 500^\circ\text{F}$$

$$s_1 = 1.5281 \text{ Btu/lb-}^\circ\text{F}_{\text{abs}}$$

At actual exhaust conditions:

$$p_2 = 2'' \text{ Hg abs}$$

$$h_2 = h_{f2} + h_{fg2} = 69.1 + (1-0.12)(1036.6) = 982 \text{ Btu/lb}$$

At hypothetical exhaust conditions after isentropic expansion:

$$s_2' = s_1 = 1.5281 \text{ Btu/lb-}^\circ\text{F}_{\text{abs}}$$

$$x_2' = \frac{s_1 - s_{f2}}{s_{fg2}} = \frac{1.5281 - 0.1316}{1.8481} = 0.757$$

$$h_2' = h_{f2} + x_2' h_{fg2} = 69.1 + 0.757(1036.6) = 788 \text{ Btu/lb}$$

Turbine efficiency is then:

$$\eta_{\text{tur.}} = \frac{\Delta h_{\text{actual}}}{\Delta h_{\Delta s=0}} = \frac{1245.1 - 982}{1245.1 - 788} = 79.5\%$$

(b) Rankine Cycle Efficiency:

$$\eta_{RC} = \frac{h_1 - h_2 - (h_4 - h_3)}{h_1 - h_4}$$

$$\eta_{RC} = \frac{(1245.1 - 788) - (70.2 - 69.10)}{1245.1 - 70.2} = 38.8\%$$

(c) Actual cycle thermal efficiency:

$$\eta_t = \frac{\Delta h_{\text{turbine, actual}}}{\Delta h_{\text{boiler}}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{h_1 - h_4}$$

$$\eta_t = \frac{(1245.1 - 982)}{1245.1 - 57.99} - 1.2 = 22.1\%$$

Availability is a useful property of a fluid that can be used in the analysis of heat power cycles. This term is defined as,

$$b = h - T_0 s$$

where, b is the availability, Btu/lb; h is the enthalpy, Btu/lb;

T_0 is the absolute temperature of the sink, F_{abs} ; and s is the entropy, Btu/lb- $^{\circ}F_{\text{abs}}$. For isentropic expansion through a prime mover, where $ds = 0$, the change in enthalpy is equal to the change in availability, but for the actual turbine expansion process, (ignoring changes in potential energy due to elevation changes, and kinetic energy due to velocity change) the change in availability is:

$$\Delta b = b_1 - b_2 = h_1 - h_2 - T_0(s_1 - s_2)$$

Fig. 3. Rankine and actual steam cycles.

Fig. 4. Typical exhaust losses of condensing steam turbines. Numbers on curves are exhaust annuli in sq ft, although there are also differences due to blading design, etc. The exhaust losses at low velocities also include the extra loss due to areas becoming disproportionately large compared to flow. (From Warren and Knowlton, op. cit.)

Since s_2 is greater than s_1 , Δ_b is greater than Δ_h . The ratio of Δ_h to Δ_b is termed the effectiveness. Comparison of the changes in availability and effectiveness of various cycles is often more illuminating as to why one cycle is better than another than in simple comparison of thermal efficiencies.

Example 2 -- What is the change in (a) availability, and (b), the effectiveness of the steam turbine in Example 1, taking the turbine exhaust temperature as the sink temperature?

$$(a) \Delta_b = h_1 - h_2 - T_0(s_1 - s_2)$$

$$s_2 = s_{f_2} + x_2 s_{fg2} = 0.1316 \times 0.78(1.8481) = 1.757 \text{ Btu/lb-}^\circ\text{F}_{\text{abs}}$$

$$\Delta_b = 1256.1 - 982 - 560.7(1.528 - 1.757)$$

$$\Delta_b = 491.4 \text{ Btu/lb}$$

(b) Effectiveness:

$$= \frac{\Delta h_{\text{actual}}}{\Delta b_{\text{actual}}} = \frac{363.1}{491.4} = 74\%$$

Gas cycle efficiencies are defined in the same way as for vapor cycles. The range of temperatures and pressures in gas cycles is within the applicable range of the perfect gas equations, so elementary thermodynamic relationships may be conveniently used with very little error.

Provided there is little pressure loss due to flow, heat additions or removal can be assumed to take place at constant specific heat so that: $Q = c_p(t_1 - t_2)$, where $Q = \text{Btu/lb}$, c_p is constant pressure specific heat $= \text{Btu/lb-}^\circ\text{F}$, and $t = \text{degrees F}$.

The work of isentropic compression is generally expressed as:

$$W = \frac{k}{k-1} RT \left[\left(\frac{P_1}{P_2} \right)^{\frac{k-1}{k}} - 1 \right]$$

Where: W = work, ft-lbs/lb

$$R = \text{gas constant} = \frac{c_p - c_v}{778.3} = \text{ft-lbs/lb-F}_{\text{abs}}$$

T_1 = temperature at beginning of process, F_{abs}

P_1 = higher pressure, lbs/sqft

P_2 = lower pressure, lbs/sqft

$$k = \text{ratio of specific heats} = \frac{c_p}{c_v}$$

c_p, c_v = constant pressure and constant volume specific heats respectively, Btu/lb-F

Using the expression for turbine efficiency, η_t , and compression efficiency, η_c , and the point designations shown on Fig. 2, more convenient forms for solution of gas turbine cycles employing different gases are:

$$W_{\text{actual turbine}} = \eta_t c_p T_1 \left[1 - \left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} \right] = \text{Btu/lb}$$

$$W_{\text{actual comp}} = \frac{c_p T_3}{\eta_c} \left[\left(\frac{P_4}{P_3} \right)^{\frac{k-1}{k}} - 1 \right] = \text{Btu/lb}$$

Also for the actual expansion process:

$$W_{\text{actual turbine}} = h_1 - h_2 = c_p (T_1 - T_2)$$

$$T_2 = T_1 - \frac{W_{\text{actual turbine}}}{c_p}$$

And for the compression process:

$$T_4 = T_3 - \frac{W_{\text{actual comp.}}}{c_p}$$

The optimum intercooler pressure, p_1 , exists when $p_1 = \sqrt{p_1 p_2}$.

Actual heat power cycles can but approach the efficiency of the Carnot cycle because heat cannot be added and rejected at constant temperature, nor can the machines be made adiabatic and frictionless. The relatively high constant temperature of heat addition during the boiling process in a vapor cycle, see Fig. 14, is a distinct thermodynamic advantage of this type of cycle. Improvements made to a cycle by reducing heat and friction losses and by lowering the sink temperature, quickly reach the point where further improvement is not economically justified. The most promising approaches are through raising the temperature of the heat source and through improving the over-all average temperature at which the heat is added. The maximum temperature is limited only by the characteristics of the materials of construction. The average temperature for heat addition is improved through (a) heat recovery, (b) regeneration, (c) reheat, and (d) multi-fluid cycles.

(a) Heat Recovery. The temperature of the working fluid may be raised before it is heated by the primary source by heat exchange with let-down streams, exhaust gases, etc., to the extent that the cost of the transfer surface is justified.

(b) Regeneration. In the single-phase fluid type cycle, regeneration consists of exchange of heat between gases after expansion through the prime mover and the compressed gas flowing back to the heat source. In the vapor cycle, steam is extracted from the turbine after partial expansion to heat the water fed to the boiler, the number of stages of feedwater heating being determined by a comparison of the investment costs against savings, the optimum usually falling between four and eight stages for large modern steam station. The relative effectiveness of adding stages of feedwater heating is shown in Table 1.

Table 1

Theoretical Effectiveness of Regeneration in
Steam Cycle*

<u>Number of Feedwater Heaters</u>	<u>Total Percent Heat Saved</u>
1	5.5%
2	7.5
3	9.2
5	10.5
10	11.2

* Based on full-load operation, 600 psia and 600°F to throttle, 1" Hg abs exhaust.

(c) Reheat. The average temperature of heat addition may be raised by reheating of the working fluid after partial expansion through the turbine. Improvements to cycle efficiency are not as marked as those obtained for the first few stages of feedwater regeneration, and in fact, unless the reheating is accomplished at a high average temperature and the steam is heated to near its initial temperature, the gains are quite small. If the average temperature of the addition of the reheat energy is less than that for the average initial heating, then there will be a

decrease in efficiency. For the vapor cycle, an equally important advantage of reheat is the improvement in the quality of the steam in the low-pressure stages of the turbine.

(d) Binary, or multiple-fluid, cycles. Since the maximum temperature is usually limited by a combination of high pressure and high temperature, it is possible to use one working fluid having desirable properties in the high temperature ranges to give up its heat after expansion to another fluid having better low-temperature characteristics. Mercury-water cycles have been used with some success, although the high cost of mercury and its toxicity are major deterrents.

The working fluid for a single-phase type heat power cycle has been limited to air until the advent of atomic energy in that heat was supplied by a combustion process with the hot gases passing directly through a gas turbine. Inasmuch as it may be wished to recirculate the working fluid in a closed loop for radioactivity containment considerations, other gases may be practical.

Despite the continued search for a better working fluid for the two-phase type cycle, no medium has been found with more desirable characteristics than has water. However, water does have the undesirable properties of high vapor pressures at the elevated temperatures of the heat sources; a pressure less than atmospheric accompanied by high specific volume at the usual sink temperatures; a critical temperature of 705.4 F, making constant temperature addition of heat above this point impossible. Mercury, diphenyls (Dowtherm), terphenyls, and combinations of these with water in binary cycles have been given much study. Dowtherm has interesting heat transfer properties and low vapor pressures at high temperatures and may have application in the power cycle of the organic-moderated type reactor.

II. PERFORMANCE CALCULATIONS FOR STEAM CYCLES

The nuclear engineer is primarily concerned with making preliminary studies of steam cycles with the aim of optimization of the cycle insofar as it is compatible with reactor design. The following section illustrates the method and furnishes information sufficient for an analysis of adequate accuracy for most initial studies. Detailed analysis of a steam cycle is usually reserved until selection of a turbine-generator unit, at which time the manufacturer can furnish actual performance data. In making use of published material showing performance of operating units it must be cautioned that, (1) subtle differences between operating conditions may have more noticeable effect on efficiency than realized; (2), builders seldom allow publication of but the best of their operating achievements; (3), the manufacturers make changes in nozzles, blading, etc., within the framework of a given unit to meet customers' particular needs, i.e., all turbines of a given size and steam conditions made by a particular manufacturer do not necessarily have the same efficiency.

Stage efficiency of a steam turbine is defined as the ratio of the actual enthalpy drop across the turbine stage to the enthalpy drop if the expansion were isentropic from the same initial conditions and for the same pressure drop. A large turbine is a series of relatively independent stages, and stage-by-stage studies are used by turbine designers to evaluate the performance characteristics of a given machine. However, the actual behavior of turbines shows a marked consistency when stages are considered by groups rather than individually, so that the much less tedious method of using overall values is sufficiently accurate for most power cycle studies.

The expanding steam in a turbine furnishes the energy for the net output at the generator terminals, energy for all electrical and mechanical losses in the generator and the turbine, and imparts a kinetic energy to the leaving stream. This net energy delivered to the turbine shaft is sometimes known as wheel output. Since heat losses from the turbine casing are small, this work is essentially the only energy taken from the steam, the remainder going into the exhaust. For a straight (non-extraction) type turbine, the initial enthalpy minus the heat converted to wheel output gives a fictitious exhaust enthalpy, known as the exhaust point (EP). If the kinetic energy, known as the exhaust loss, is subtracted from the exhaust point enthalpy, the state-line end point (SLEP) obtained, which is more nearly representative of the actual enthalpy in the exhaust and is useful in construction of the expansion curve for the turbine.

Turbine internal efficiency, or expansion line efficiency, or Rankine cycle ratio, is the ratio of the actual energy delivered to the turbine shaft and to the leaving stream, to the available energy from isentropic expansion from the same initial conditions at the turbine stop valve to the same exhaust pressure.

$$\eta_{\text{internal}} = \frac{h_1 - h_2(\text{SLEP})}{h_1 - h_2}$$

The values in the above expression are as shown in Fig. 3, with the value of " h_2 " being the state-line end point referred to above.

Turbine efficiency, or engine efficiency, is the ratio of the actual energy delivered to the turbine shaft not including the exhaust kinetic

energy, to the available energy from isentropic expansion from the initial conditions at the turbine stop valve to the same exhaust pressure.

$$\eta_{\text{turbine}} = \frac{h_1 - h_2 (\text{EP})}{h_1 - h_2}$$

The turbine efficiency, as does the internal efficiency, takes into account the pressure losses due to flow through the stop valve, throttle valve, governing valves, nozzle losses, flow friction and windage losses, bearing friction, oil pump power, friction and electrical losses in the generator, and exciter power.

If the turbine is a straight type (non-extraction) the turbine efficiency is also:

$$\eta_{\text{turbine}} = \frac{3,413}{W_a (h_1 - h_2)}$$

where: W_a is the actual lbs/hr steam to the turbine throttle per kw delivered at the generator terminals; a value known as the actual steam rate. The theoretical steam rate, W_t , is the amount of steam required, lbs/hr, to generate 1 kw if the expansion were isentropic and there were no losses in the system.

$$W_t = \frac{3,413}{h_1 - h_2}$$

$$\text{then, } W_a = \frac{W_t}{\eta_{\text{turbine}}} = \frac{3,413}{\eta_{\text{turbine}} (h_1 - h_2)}$$

The magnitude of the exhaust loss varies with the area of the exhaust annulus. Various areas may be used by a manufacturer for a given turbine rating depending on the efficiency for which the customer is willing to pay, since a large annulus decreases the exhaust loss but increases the manufacturing cost. Typical annuli are 1 to 2 sq ft/1000 kw rated capacity. It is assumed that if it were possible to utilize the leaving loss it would be used with an efficiency equal to that of the turbine internal efficiency; hence, exhaust losses should be multiplied by the internal efficiency when locating the state-line end point. The exhaust loss on a percentage basis is the exhaust loss, in Btu/lb, divided by the available energy change between the steam conditions ahead of the stop valve and the exhaust pressure. Typical exhaust losses are about 4%. The exhaust loss varies somewhat with load, but since the variations are but a small part of the total, preliminary studies neglect corrections to exhaust loss for partial load conditions. Exhaust losses for typical turbines are shown in Figure 4.

The above definition of turbine efficiency was based on a non-extraction type turbine, but the term may also be applied to extraction units. Numerical values of turbine efficiency are higher for the straight type because of the additional losses imposed by extraction, particularly automatic extraction, so that values of turbine efficiency for a straight unit do not apply to those for the same size unit arranged for extraction. In making steam power studies at least some extraction will probably be involved, and in this case, the enthalpy at the extraction point must be approximated in order to make energy balances. The easiest method of estimating the enthalpies is to construct on the $h-s$ plane, or Mollier diagram, the expansion line, or condition curve representing the loci of

conditions entering and leaving the various stages. This line does not represent the continuous change in the properties of the steam as it goes through the turbine stages, however.

Manufacturers can supply actual performance data on a particular machine, but in making preliminary studies it is usually sufficient to construct a typical expansion curve. It is sometimes adequate to simply draw a straight line on the $h-s$ plane between the conditions at the stop valve and the calculated exhaust point enthalpy, see Figure 5. A better method is to estimate the throttle end pressure losses due to stop and throttle valves, strainer, and governor (normally 3 to 5% of the initial pressure) and to locate the start of the expansion line at this point after constant enthalpy expansion (throttling loss) to the corrected pressure entering the first stages.

A more exact representation is a curved line having a greater slope in the first stages of the turbine than in the latter stages, which have some retardation of efficiency due to moisture in the steam. Enthalpies needed for energy balances are almost always taken from the lower portion of the curve so that its exact shape is not of prime importance. A graphical method of construction of more nearly representative expansion curves has been devised by Warren and Knowlton*, as shown in Fig. 5 and described below:

The exhaust point enthalpy, h_{2-EP} , is calculated from an estimated value for the turbine efficiency. The exhaust loss is estimated from

* Warren and Knowlton, "Relative Engine Efficiencies Realizable from Large Modern Steam Turbine-Generator Units", Transactions, Am. Soc. Mech. Engrs., 1941.

Fig. 5. Method of construction of steam turbine condition curve on Mollier Diagram. (From Warren and Knowlton, op. cit.)

available data, such as Fig. 4 of by assuming a value of 4%, expressed in Btu/lb, and multiplied by the turbine internal efficiency, which can be approximated closely enough by assuming it to be about 1.05 to 1.10 times the turbine efficiency. This value for exhaust loss, corrected for internal efficiency, is then subtracted from h_{2-EP} to give the state-line end point, h_{2-SLEP} . A straight line is then drawn between these two points. The approximated expansion curve can be assumed to cross this line at point "C", 1/4th of the distance from A to B, the expansion curve being slightly above it from A to C, as shown. The curve will fall below the line by the equivalent of about 6 Btu/lb at the point, "D", half-way between points B and C. The approximate expansion line can then be drawn in through the four approximated points, ABCD.

If the turbine is supplied with saturated steam to throttle, the expansion curve will probably fall below the straight line for the entire path and can be more closely approximated by assuming that it resembles the "C" to "B" portion of the curve shown in Fig. 5.

The turbine efficiencies realizable from typical units depend upon the design throttle conditions, exhaust pressure, speed, and size of the unit. For a given machine, operation at partial load, or with other than design conditions at throttle or exhaust, will have marked effect on the turbine efficiency. Some typical turbine efficiencies are shown in Fig. 6 and correction factors are indicated in Figs. 7 and 8.

The values shown in Fig. 6 are turbine efficiencies for units having 300°F superheat at throttle. While a correction factor can be approximated from Fig. 8 for saturated steam to throttle, the design and arrangement of the unit may be affected so much that manufacturer's

Fig. 6. Turbine efficiencies for condensing turbine-generator units.

Based on 300 F initial superheat, 4% exhaust loss, 1½% mechanical loss at all ratings, generator efficiencies between 98 and 99% with hydrogen cooling. Numbers on curves are throttle pressure, psig; full lines are for 1800 rpm units and dashed lines are for 3600 rpm units. (From Warren and Knowlton, *op. cit.*)

Fig. 7. Partial load correction factor for condensing steam turbines. Full load turbine efficiencies are to be multiplied by this factor. Numbers on curves are volumes of steam flow at exhaust conditions. (From Warren and Knowlton, op. cit.)

Fig. 8. Superheat correction factor for condensing steam turbines. Turbine efficiencies taken at 300 F of superheat (Fig. 3.6) are to be divided by this factor. (From Warren and Knowlton, op. cit.)

data would be considerably more reliable. For estimating purposes, values of internal turbine efficiency of large two-element saturated steam turbines are shown in Figs. 9 and 10. Knowing the internal efficiency allows quick estimate of the state-line end-point. To determine the actual work output of the turbine using these values, an exhaust loss of about 20 Btu/lb may be assumed.

Mechanical losses in a turbine do not vary with load, so the percentage of mechanical loss varies inversely as the load on the machine.

Fig. 9. Internal turbine efficiency of low-pressure turbine elements using saturated steam. (From Beldecos and Smith, op. cit.)

Fig. 10. Internal turbine efficiency of intermediate and high-pressure 3600 rpm turbine elements using saturated steam. (From Beldecos and Smith, op. cit.).

$$\text{PRESSURE RATIO} = \frac{(\text{Throttle, psi})}{(\text{Exhaust, psi})}$$

Example 3

A steam turbine of 100-mw capacity, with a cycle arrangement as shown in Fig. 15, is supplied with steam at 2400 psia and 1000°F. It exhausts at 1.5" Hg abs. What is estimated cycle thermal efficiency?

See Fig. 11.

a. Isentropic Δh_s from initial conditions to exhaust pressure:

$$x_{h_1} = \frac{s_1 - s_{f_{h_1}}}{s_{fg_{h_1}}} = \frac{1.5335 - 0.1147}{1.8894} = 0.751$$

$$h_{h_1} = h_f + x_{h_1} h_{fg} = 59.7 + 0.751(1042) = 842.2 \text{ Btu/lb}$$

$$\Delta h_{1-h_1} = 1461.7 - 842.2 = 619.5 \text{ Btu/lb}$$

b. Turbine efficiency, from Fig. 6 = 75.9%.

Correction factor for 338°F superheat, from Fig. 8 = 0.995.

$$\text{Corrected overall turbine efficiency} = \frac{75.9\%}{0.995} = 76.0\%$$

c. Exhaust loss assumed as 4%.

$$\text{Exhaust loss} = 0.04 \times \Delta h_s = 0.04 \times 619.5 = 24.8 \text{ Btu/lb.}$$

$$\text{d. Internal efficiency} = \frac{(619.5 \times 0.76) + 24.8}{619.5} = 80.0\%$$

$$\text{e. State-line end point} = h = 1461.7 - 0.80(619.5) = 966.1 \text{ Btu/lb.}$$

f. Assume loss through throttle valve as 4%.

$$p_A = 2400 \times (1 - 0.04) = 2304 \text{ psia}$$

g. Condition curve can now be constructed through points 1, A, and B, see Fig. 5 and Fig. 11.

h. Assume first heater deaerating type with water leaving at 212°F.

Assume 5 psig heating steam, then $p_3 = 20$ psia; and $h_3 = 1120$ Btu/lb, from Fig. 11.

- i. Assume heating from outlet of open heater to saturation temperature in boiler to be roughly equally divided between boiler and heater:

$$t_9 = \text{say, } 400^{\circ}\text{F}$$

- j. Boiler feedpump work, assuming efficiency of 75%:

From compressed liquid tables:

$$\Delta h_p = \frac{\Delta h_s}{\eta} = \frac{4.5}{0.75} = 6.0 \text{ Btu/lb}$$

$$h_9 = 376 + 2.66 = 378.7 \text{ Btu/lb}$$

- k. Assume terminal temperature difference in closed heater of 50°F .

$$t_2 = 450^{\circ}\text{F, and } p_2 = 422.6 \text{ psia}$$

$$h_2 = 1315 \text{ Btu/lb, from Fig. 11.}$$

- l. Extraction steam required for closed heater, w_2 :

$$w_2(1315.0 - 430.1) = (378.7 - 186.1)$$

$$w_2 = 0.1614 \text{ lbs/one-lb to throttle}$$

- m. Extraction steam required for open heater, w_3 :

$$h_3 w_3 + j_6(1 - w_2 - w_3) + h_{10} w_2 = h_7$$

$$1120 w_3 + 59.7(0.8386 - w_3) + 430.1(0.1614) = 180.0$$

$$w_3 = 0.0571 \text{ lbs/one-lb to throttle}$$

- n. Turbine work output:

$$W = w_1(h_1 - h_2) + (1 - w_2)(h_2 - h_3) = (1 - w_2 - w_3)(h_3 - h_4)$$

$$W = (1461.7 - 1315) + 0.8386(1315 - 1120) + 0.7815(1120 - 991) =$$

$$W = 411.2 \text{ Btu/lb}$$

- o. Net work = Work turbine - work for pump (step j)

$$W_{\text{net}} = 411.2 - 6.0 = 405.2 \text{ Btu/lb}$$

- p. Heat added in boiler:

$$Q_{\text{in}} = h_1 - h_9 = 1461.7 - 378.7 = 1083 \text{ Btu/lb}$$

g. Thermal efficiency of cycle:

$$\eta_t = \frac{W_{\text{net}}}{Q_{\text{in}}} = \frac{405.2}{1083} = 37.4\%$$

SECTION III. - PERFORMANCE CALCULATIONS FOR GAS TURBINE CYCLE

Gas turbines are normally associated with the burning of oil or gas in combustion chambers from which hot gases discharge directly through turbine blading. Units of this type employ temperatures of 1400⁰F or higher, but at pressures seldom over 100 psig. The simple gas turbine cycle employs a centrifugal compressor, a means of heating the compressed gas, and a turbine to drive both the compressor and the electrical generator, such as shown in the open cycle, Fig. 12. Heat recovery devices, multi-stage compression with intercooling, and regeneration, as shown in Fig. 13, all improve the thermal efficiency of the cycle. The relatively poor gas film heat transfer makes heat exchanger surfaces relatively expensive, however, so that the economics of the cycle must be carefully investigated.

Turbine efficiencies generally run higher than for steam turbines because of the absence of moisture in the low-pressure stages. Extensive data on large gas turbines for power plant applications is not available since practically all installations have been under 10,000-kw electrical generating capacity. The turbine efficiency for large size units may be assumed to fall between 85 and 89%. Compression efficiencies are between 82 and 83% for similar-sized units. About 60 to 70% of the turbine output power is required to drive the compressor. Leaving losses are somewhat higher in the gas turbine than for steam turbines.

The pressure ratio in a gas turbine cycle is the ratio of the highest compressor discharge pressure to the lowest compressor inlet pressure. The efficiency of the ideal Brayton cycle, which consists of simple isentropic compression and expansion processes and constant pressure heating and cooling, increases as the cycle pressure ratio is increased. The performance of

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Fig. 12. Simple open gas turbine cycle.

Fig. 13. Closed gas turbine cycle with regeneration, precooling and intercooling. See Example 3.4.

actual machines is greatly different from that of the theoretical cycle, however, so that an optimum pressure ratio exists. Diminishing returns on investment cost of heat transfer surfaces also dictates an optimum ratio, which usually is not over about four. Such optimization studies are often most conveniently made through trial calculations, such as in Example 4.

The open gas turbine cycle has simplicity to commend it, but atmospheric air has relatively poor heat transfer properties, has a high pumping cost, is corrosive at high temperatures, and becomes somewhat active due to formation of argon-41. A closed cycle permits use of gases other than air as the working fluid in order to better the efficiency and to control corrosion, but involves the use of a precooler to cool the turbine exhaust prior to re-entry into the compressor, and also dictates close attention to the problem of radioactivity build-up in the system. Hydrogen would be a good working fluid in that it has good heat transfer characteristics, and requires low pumping effort, but the materials in the system must be selected to withstand attack or embrittlement by hydrogen, and further, it also introduces an explosion hazard. The gases that have received most serious consideration for closed cycles are helium, nitrogen and carbon dioxide. Helium is inert to most materials and has a very low cross section for neutron capture. It is relatively expensive, however, and leaks are difficult to control. Nitrogen exhibits about the same thermodynamic behavior as does air, and is inert to graphite and stainless steels up to about 1800°F (in the absence of oxygen). The (n,p) reaction however, forms C^{14} which constitutes a hazard from beta emissions if it escapes from the system. The carbon and oxygen nuclei in carbon dioxide have low cross sections for neutron absorption and the gas does not readily attack most reactor materials. Its nuclear and heat transfer

properties and the required pumping effort are not as good as for helium, but it is much more economical. Carbon dioxide is safe in regards to toxicity and explosion hazards. It may react with graphite at higher temperatures, and its interaction with other materials in intense radiation fields needs further investigation.

Parasitic pressure losses must be minimized in the system because of the large fraction of turbine work that must be used for the compression process. Pressure losses through heat transfer equipment are generally assumed at 1.5% to 4% of the entering pressure. Increased cycle temperatures reduce the sensitivity to pressure losses.

Example 4. A closed gas turbine cycle is arranged as shown in Fig. 13, and uses helium as the working fluid. Turbine efficiency is 85% and compression efficiency is 83%. The maximum temperature in the cycle is 1000°F and the minimum temperature is 92°F. What is the optimum pressure ratio, and the thermal efficiency of the cycle?

Properties of helium are:

$$c_p = 1.25 \text{ Btu/lb-}^\circ\text{F}$$

$$k = \frac{c_p}{c_v} = 1.659$$

For a pressure ratio ($n = \frac{p_1}{p_2}$) of 1.75:

$$W_t = 1.25 \times 1460 \times 0.85 \left[1 - \left(\frac{1}{1.75} \right)^{\frac{1.659-1}{1.659}} \right] = 308.9 \text{ Btu/lb}$$

For two-stage compression, with perfect intercooling, i.e., $T_6 = T_4$:

$$n_1 = \frac{p_5}{p_4} = \frac{p_7}{p_6} = \sqrt{\frac{p_1}{p_2}} = \sqrt{n}$$

$$W_c = \frac{2 \times 1.25 \times 552}{0.83} \left[\left(\sqrt{1.75} \right)^{\frac{1.659-1}{1.659}} - 1 \right] = 195.2 \text{ Btu/lb}$$

$$W_{\text{net}} = 308.9 - 195.2 = 113.7 \text{ Btu/lb}$$

The temperature leaving the turbine, T_2 , is:

$$T_2 = T_1 - \frac{W_t}{c_p}$$

$$T_2 = 1460 - \frac{308.9}{1.25} = 1212.9$$

$$T_8 = 0.92 \times 1212.9 = 1115.9^\circ \text{F}_{\text{abs}}$$

The temperature, T_8 , leaving the regenerator will depend on the investment for heat transfer surface. In this example it is arbitrarily assumed that T_8 is 92% of T_2 , but it must be borne in mind that the calculated efficiency will be sensitive to this assumption.

$$Q_{\text{add}} = c_p(T_{11} - T_8) = 1.25(1460 - 1116) = 430 \text{ Btu/lb}$$

The thermal efficiency is:

$$\eta_t = \frac{W_{\text{net}}}{Q_{\text{add}}} = \frac{113.7}{430} = 26.4\%$$

If other pressure ratios are substituted, and T_8 is always assumed as 92% of T_2 :

$n = \frac{P_1}{P_2}$	W_t	W_c	W_{net}	Q	η_t
1.50	162.1	139.7	22.4	295.1	7.6%
1.75	308.9	195.2	113.7	430.1	26.4
2.00	373.9	245.2	128.6	489.9	26.2
3.00	549.1	405.7	143.4	651.2	22.0

The most efficient pressure ratio is thus indicated to be about 1.75.

IV. COMPARISON OF THE STEAM AND GAS CYCLES

There is an obvious thermodynamic advantage to using the power cycle working fluid to cool the reactor itself rather than taking the heat from an intermediate heat transporting fluid, since the temperature difference in the heat exchanger represents a serious loss in availability of energy. Gas-cooled and boiling water reactors are of this type. Both of these assume that leakage of the working fluid can be controlled and that activity containment problems can be met with reasonable cost. Consideration of higher temperatures leads toward preference for the gas turbine cycle because of the elevated pressures associated with high temperature water. Since conventional gas cycles operate with maximum pressures under 100 psig, temperatures of 1400^oF and higher have been obtainable as compared to a 1150^oF maximum for steam cycles. Heat transfer considerations in gas-cooled reactors, however, point up need for gas pressurization in order to achieve better heat transfer coefficients, and some gas cycles have been proposed with pressures as high as 1,000 psig.* This factor would tend to negate one of the primary advantages of the gas cycle, particularly in that the entire system would have to withstand the high pressure whereas the exhaust system of a steam cycle is always near atmospheric pressure.

Gas turbines for stationary power plants have not been manufactured in sizes larger than 30,000-kw, so that extension of nuclear power plant studies into areas of turbine sizes comparable to steam plants is largely speculative, both as to investment and maintenance costs. It is clear that unless temperatures of over 1200 to 1500^oF are attainable, the gas turbine cycle appears

* Daniels, Farrington, "Small Gas-Cycle Reactor Offers Economic Promise", Nucleonics, March, 1956, pages 34-41.

to hold little promise*. Efforts to use high temperatures have led to special turbine blading problems. Progress has been made to where blade lives can be expected to compare equally with those of a steam turbine, particularly in that the gas turbine blading is not subject to the moisture erosion of the steam blading. Use of very high temperatures also implies reactor component temperatures of even higher magnitude, perhaps beyond the range of metallic fuel plates. Ceramic fuel elements have long been considered attractive because of neutron economy, higher permissible working temperatures, and easier processing and fabrication. Poor resistance to thermal stress, and in particular, the probability that abrasive particles will be swept through the turbine blading, has caused some skepticism because of the vulnerability of blading in these respects.

Whether the power cycle be either the steam or the gas turbine type, there is a great advantage to lowering of the temperature of heat rejection. The steam condenser of the former and the precooler of the latter may be either air or water cooled. Air cooling has the obvious advantage of freeing the plant from dependence on large volumes of cooling water, and in many locations in the U. S., the year-around average air temperature is not greatly above that of the available water supply. Because of the wider fluctuations of air temperature, however, if the installed capacity is sufficient to meet power demands during warm days when the thermal efficiency is reduced, the equipment will operate at low load factors on cool days. The possibility of freezing condensate in an air-cooled steam condenser cannot be neglected. The advantage also lies with water-cooled heat rejection equipment for both cycles in regards to heat transfer coefficients and surface area needed.

The open gas turbine cycle has promise for plants at remote locations,

* "Nuclear Gas Turbines", Mechanical Engineering, July, 1956, page 606.

or for transport equipment, in that the equipment can be compact, lighter in weight, relatively easy to control, operate with less loss in efficiency at partial loads, and is less dependent on having a source of cooling water. These factors are not as important for large stationary power plants having good load factors, however, where a premium is placed on high plant efficiency.

Since about 60 to 70% of the gas turbine power output is required to drive the centrifugal compressor, the gas turbine must have a horsepower output about one and one half times as great as for a steam turbine to drive an electrical generator of equal output. This not only makes for relative greater investment cost in the machine, but since one is depending on the difference between turbine output and compressor requirement, only minor changes in efficiency or increased pressure loss due to flow, can cause rapid drop off of the net output.

If the steam cycles are compared on the T-s plane, see Fig. 14, with both diagrams drawn to the same upper and lower limits of absolute temperature, it may be readily seen that the enclosed area (which represents heat converted to useful work) is larger for the steam cycle. This fact is due to the constant temperature boiling and condensation processes which give opportunity for higher average temperatures of heat addition and rejection. Table 2 shows calculated values for a gas turbine cycle, arranged as shown in Fig. 13 and operating at 1000°F and 1400°F maximum temperatures, compared to a steam cycle, arranged as shown in Fig. 15 and operating at 800°F and 1000°F maximum temperatures; all cycles operating with the same minimum sink temperature of 92°F.

Fig. 14. Comparison of the gas turbine cycle and the steam cycle on the T-s plane.

Fig. 15. Simple superheated steam cycle with two stages of feedwater heating. See Example 3.3.

Table 2

Theoretical Thermal Efficiencies Gas Turbine and Steam Cycles

Max. temp.	<u>Steam Cycle*</u>		<u>Gas Turbine Cycle**</u>				
	800°F	1000°F	c_p	k	n^{***}	1000°F	1400°F
Water	31.7%	37.4%	-	-	-	-	-
Helium	-	-	1.25	1.659	1.85	26.4%	37.0%
Nitrogen	-	-	0.248	1.40	2.5	26.4%	36.9%
Carbon dioxide	-	-	0.202	1.29	3.0	26.5%	36.8%

* Superheated steam cycle as shown in Fig. 15; initial pressure 2400 psia at 1000°F, and 1600 psia at 800°F; turbine efficiency of 80%, feedwater pump efficiency of 50%; exhaust pressure 1.5" Hg abs.

** Gas turbine cycle as shown in Fig. 13; temperature T_3 assumed 92% of T_2 ; turbine efficiency of 85%; compressor efficiency of 83%, $T_4 = T_6 = 92^\circ\text{F}$. Pressure loss thru heat exchangers and reactor have been neglected.

*** Approximate pressure ratio for maximum cycle efficiency as determined by trial solutions. This ratio will be affected by assumed values of T_3 .

V. SATURATED STEAM CYCLE PERFORMANCE AND DESIGN CONSIDERATIONS

While the promise for the future may be great, the nuclear power plant in the present state of the art is often limited to temperatures for steam generation well below those commonly used in conventionally-fired plants. Superheating is difficult to accomplish because of heat transfer and hold-up problems, and unless the superheat is more than about 100°F, a saturated cycle with moisture separation may be equally as efficient and practical*.

Attendant to use of lower enthalpy steam is the higher mass flow rate required through the turbine for a given power output. Coupled with this factor is the higher specific volume of the steam at the lower pressures, indicating that for a given capability the low-pressure turbine must be larger and cost more per kilowatt of capacity. Turbine size is limited by the exhaust annulus flow area attainable; this area is established by the permissible length of the blading. Blade lengths are limited to a maximum of about 30" since they are subjected to high bending stresses, high tip speeds, mechanical vibration, etc., yet must not change shape appreciably because of the small clearances between moving and stationary blades. For a given tip speed, the blades for 1800 rpm machines can be longer than those for 3600 rpm units, so as a result the former can be made to take greater mass flows. The limitations on the amount of exhaust annulus area attainable requires compounding of the low pressure ends. The large volume of steam at the throttle end of a turbine used with saturated

* Beldecos, N. A., and Smith, A. K., "Comparative Performance of Turbine-Generator Units in Saturated Steam Cycles", Paper No. 54-SA-65, Am. Soc. Mech. Engrs.

steam makes the governing valves and operators more costly. The total number of stages required for the saturated steam turbine is of course less than for those designed for greater total enthalpy drops.

The detrimental effects of entrained moisture in the steam flow through turbine blading are well known. The useful life of the low-pressure blading is estimated at about 10 years for 12% moisture and about 6 years for 15% moisture. Stellite inserts at points of greatest erosion and stainless steel turbine cylinder liners have been used with some success. The reduction of the turbine efficiency by moisture may be roughly approximately as 1.0 to 1.2% reduction for every 1% moisture in the turbine exhaust. Since the amount of moisture at the exhaust increases as the throttle pressure is increased, turbine efficiencies drop off with increased steam pressure, as indicated in Figures 9 and 10.

The exhaust loss is a correspondingly greater percentage of the total enthalpy drop through the turbine, since the total available energy is less using saturated steam. The exhaust loss for large saturated steam turbines may be assumed at about 20 Btu/lb, corresponding to an annulus velocity of about 1000 ft/sec.

The Moisture Problem & Methods of Reduction

When saturated steam is expanded to a low pressure in a turbine, the quantity of moisture in the last blades becomes impractical (over 12 to 14%) when throttle pressures are increased to over about 50 psig. It is readily apparent that use of higher exhaust pressures to reduce moisture is not logical, since the available energy per unit of pressure drop increases markedly as the lower exhaust pressures are approached. As will be covered subsequently, raising of the exhaust quality through lowering

of the throttle pressure by throttling (to attain superheat) is also impractical.

Moisture may be controlled in a turbine through use of reheat after partial expansion, or by means of moisture separators.

Reheat may be furnished by: (1) conventionally-fired furnaces, (2) reactor heat, (3) heat transfer from saturated steam at initial throttle conditions. Method (1) is not discussed further because if the complexity of a separate combustion-type plant is added to the atomic plant, it is generally best utilized for initial superheating. Method (2), involving use of reactor heat, requires relatively large heat transfer surfaces and large shielding, piping and equipment to accommodate the lower-pressure steam. It is possible that as the design of reactors becomes more advanced that there will be parts of the system that could be cooled by steam, making reheat from reactor heat feasible, but present thinking regarding the need for an intermediate heat transferring fluid seems to largely discourage its use. Method (3), by which the partially-expanded steam is heated by throttle-condition steam involves considerable sacrifice of available energy in the higher-pressure steam by the irreversibility of the heating process, and while perhaps of limited application, such as in transport, this method does not appear practical for large stationary plants.

Moisture separation may be accomplished either inside or outside the turbine, or the methods may supplement each other. Slotted backs on the low-pressure blading have been used with some success. A vectorial analysis, see Fig. 16, of the velocities at the turbine blade shows that the entrained moisture, travelling at about $1/2$ the speed of the steam, cuts

Fig. 16. Velocity vector diagram for turbine blading showing how relative direction of moisture is across leading edge of blades and striking backs. Slots (exaggerated) for moisture collection are indicated.

Fig. 17 Comparative performance of saturated non-regenerative steam cycle with (a), reheat to initial temperature, (b), separation to 1% moisture, (c), reheat to 1% moisture. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op. cit.).

across the entering edge and strikes the back of the blades. Moisture tends to collect in the slots provided, and be thrown outward by centrifugal force into pockets in the turbine casing where it can be drained to a feedwater heater. These separators can remove about one-fourth of the moisture entering each stage. This diagram also illustrates the opportunity for erosion as the moisture cuts across the leading edge. Moisture separation external to the turbine is accomplished in either centrifugal types or impingement (baffle or screen) types. The centrifugal separators may be either rotative or stationary, the higher efficiency and broader range of operation of the former being offset by the lower maintenance and power requirements of the latter. Drains from the separators are used for feedwater heating. In general the external type separators are capable of reducing the moisture content to about 1% with a pressure drop of about 5% of the inlet pressure.

The reheat and moisture separator cycles are compared in Figure 17. When reheat is to initial throttle temperature, the cycle efficiency is greater than for the separator cycle, but when reheat is used to reduce moisture to 1% (the expected efficiency of the separator) the separator cycle is as good or better. Maximum efficiency is obtained when the separator is located at 10% of the throttle pressure. This efficiency peak is coincident with the separator location that results in identical amounts of moisture in both the high and low pressure turbine elements, as shown in Fig. 18.

The effect of the number of separators on non-regenerative cycle efficiency is shown in Figure 19, which assumes 1% moisture leaving the separator. Turbine outlet moisture is shown in Figure 20 for various saturated steam throttle pressures and separator locations. (Curves

Fig. 18. Effect of reheat or separator location on turbine outlet moisture with single-stage reheat or separation, for moisture reduction to 1% in non-regenerative saturated steam cycle exhausting at 1.5" Hg abs. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op. cit.)

Fig. 19. Effect of location of number of separators on non-regenerative saturated steam cycle thermal efficiency, with 1.5" Hg abs exhaust and moisture reduction to 1%. Numbers on curves are number of separators used. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op. cit.)

Fig. 20. Effect of location and number of separators on turbine outlet moisture in non-regenerative saturated steam cycle for various throttle pressures. Exhaust pressure is 1.5" Hg abs. High and intermediate pressure turbine element outlet moisture is shown by curve "X". Low pressure moisture curves are designated by number of stages of separation. Turbine efficiencies were taken from Fig. 3.9 and 3.10. (From Beldecos and Smith, op. cit.)

Fig. 21. Effect of number of separators on thermal efficiency of optimum non-regenerative saturated steam cycle with 1.5" Hg abs exhaust and separator outlet moisture of 1%. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op.cit.)

Fig. 22. Effect of number of feedwater heaters on thermal efficiency of regenerative saturated steam cycle with one and two separators. Heaters and separators are at optimum locations. Exhaust is 1.5" Hg abs. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op.cit.)

for 5% moisture at separator outlet are given in the source reference*).

As throttle temperatures are increased, the justification for additional separators becomes greater, as shown in Fig. 21 for the nonregenerative cycle and in Fig. 22 for the regenerative cycle with up to three feedwater heaters. Gain in efficiency due to regeneration is less than in conventional cycles in that separate drains are capable of substantial amounts of feedwater heating. The fewer stages in a low pressure turbine also limit the number of available extraction points on the turbine.

Since machine capability is limited by the exhaust annulus obtainable, a further advantage of the separator cycle is the reduction of the mass flow rate to the condenser, an effect shown in Fig. 23, for a tandem compound turbine-generator unit based on present day low pressure element design limitations and exhaust annulus area requirements predicated on 20 Btu/lb (at 1.5" Hg abs back pressure) exhaust loss. Regeneration of course decreases the condenser mass flow rate further.

Superheating at Expense of Throttle Pressure

For a given maximum outlet steam temperature obtainable from a power reactor there is a choice of using saturated steam or superheated steam at a lower pressure. While superheat is normally thought of as a means of increasing turbine efficiency by raising the quality of the steam in the last stages of the turbine (and also as a measure to prolong blade life), superheat gained, in effect, by throttling of the steam to a lower pressure is a degradation of energy that more than offsets the gain in turbine efficiency. As a result the cycle shows less thermal efficiency

* Beldecos, op. cit.

Fig. 23. Maximum capability of tandem-compounded turbine-generator units, based on low-pressure end exhaust annuli requirements. (From Beldecos and Smith, op. cit.)

Fig. 24. Thermal efficiency of straight condensing turbine steam cycle at various throttle conditions with 1.5" Hg abs exhaust pressure. Turbine efficiencies were taken from Fig. 3.9 and Fig. 3.10. (From Beldecos and Smith, op. cit.)

than would a saturated steam cycle. Although in the saturated cycle the moisture in the last stages of the turbine may be excessive, requiring the use of moisture separators, cost of this equipment is offset by the greater heat transfer surface and pipe sizes required for the superheat cycle, so that there appears little justification for superheat gained at the expense of throttle pressure. Fig. 24 shows typical cycle thermal efficiencies as a function of the throttle conditions, taking into account the difference in turbine efficiency due to moisture. In examining Fig. 24 it is to be remembered that a throttling process, while providing superheat at the lower pressure also lowers the temperature of the steam.

VI. THE SUPERCRITICAL PRESSURE STEAM CYCLE

While the atomic power plant may not be developed sufficiently in the next few years to make use of boiling pressures in excess of 3206.2 psia (705.4°F), the supercritical pressure steam power plant of the Ohio Power Company at Philo, Ohio, is deserving of special mention because it is one of the most significant thermodynamic developments in many years*.

In this plant the boiler feedwater is pumped into the steam generators at about 5000 psi, and as the water is heated to steam there is only a subtle change of phase from liquid to gaseous because above the critical pressure the value of latent heat of vaporization, h_{fg} , and the specific volume of vaporization, v_{fg} , are zero. At supercritical pressures the specific volume and enthalpy of the fluid gradually increase with temperature until the range of $700\text{--}800^{\circ}\text{F}$ is reached, where both then increase rapidly; above 800°F the change becomes more gradual again. Steam is delivered to the turbines at 4500 psi and 1150°F . Of the 985 Btu/lb heat added to the

* Rowand, W. H. and Frendberg, A. M., "First Commercial Supercritical Pressure Steam Generator for Philo Plant", Paper No. 55-A-135, Am. Soc. Mech. Engrs.

cycle, 235 Btu/lb are used to raise the 525°F feedwater to the critical temperature of 705°F, and the remaining 750 Btu/lb are used to superheat to 1150°F. These values compare to about 705°F Btu/lb used to form steam in a conventional steam generator and about 370 Btu/lb for superheating (at throttle conditions of 2000 psig and 1050°F with 450°F feedwater). This poses the considerable problem of arrangement of heat transfer surfaces in that most of the heat must be added at such high temperature that it is difficult to cool the exit combustion gases sufficiently to realize good thermal efficiency, not to mention the practical problems of refractory life, corrosion of boiler tubes, and the desirability of keeping air heater temperatures below 800°F.

Since there is no boiling in the conventional sense, the usual thick-walled upper boiler drums with internals for releasing steam from the water with low carry-over of moisture and solids, and the equally expensive lower, or "mud", drum, used for accumulation of solids for blowdown, can be eliminated with considerable savings. Since the flow through the supercritical boiler is "once-through", solids in the water are either carried out with the steam or deposited on the tube surface. European experience with this type of steam generation indicated that only extremely pure water of less than 0.5 ppm (as compared to allowable solids of 1 to 2 ppm for conventional units) could circumvent this problem. However, the excellent solvent action of high temperature pure water not only attacks the boiler tubes themselves but the silicas and sulphates normally deposited in the boiler tubes are also soluble and are carried out in solution with the steam to be deposited on the turbine blades when the steam has expanded to more "normal" conditions where the

solids are practically insoluble.

55.

There are many other interesting aspects of this pioneer plant, but the salient fact is that the plant will have a thermal efficiency of over 40% (61% perfection) as compared to about 37% (57% perfection) for conventional present-day large steam stations.

VII. HEAT POWER CYCLES FOR ATOMIC PLANTS

In the present stage of development, heat power cycles are largely tailor-made to fit a particular reactor concept. There are a large number of different reactor designs under construction and development, so it is probable that there will be no clear cut picture of the most promising cycles for atomic power plants for some time. The heat-power cycle conditions for the large nuclear power plants planned to date are shown in Table 3.

General Electric "Dual Cycle"*

The Dresden Nuclear Power Station has a rated capacity of 180,000 kw (with circulating water at 75°F) and is a boiling water type reactor. Water is used as the moderator and since steam voids reduce reactivity, the proportion of steam to water can be used to control power output. The so-called "dual cycle" provides a method for heat removal at various loads without exceeding the maximum allowable steam volume in the core. Briefly, this is accomplished by taking the 1000 psia saturated water from the boiling reactor and pumping it through a secondary boiler which generates saturated steam at 500 psia, the cooled water being returned to the primary boiler. The cycle is shown schematically in Fig. 25.

* Elliott, Vorrar A., "180,000-kw Dresden Nuclear Power Station", Atomic Power Equip. Dept., General Elec. Co., Paper presented in Philadelphia, Pa., Jan. 30, 1956.

Table 3

Power Plant Data for Proposed Large Atomic Power Stations*

Company & Location	Reactor Type	Reactor Power mw	Generating Capacity mw	Est. Cost Turbine Plant, \$	Est. Total Plant Cost \$/kw cap.	Est. Power Cost mills/kwh	Steam Conditions	Steam Flow lbs/hr
Consolidated Edison, Indian Point, N.Y.	Pressurized Water	500	236 ⁽¹⁾	--	230	9	370 psi	1,900,000
Commonwealth Edison, near Chicago	Boiling Water	625	180	\$10,800,000	250	-	1000 psi 545°F	--
Yankee Atomic Elec., Rowe, Mass.	Pressurized Water	480	134	\$16,000,000	246	-	600 psi 486°F	
Pa. Power & Light, Service Area	Homogeneous		150				600 psi	
Duquesne Light Co., Shippingport Pa.	Press. Water	236	60	\$10,000,000	370		585 psi saturated	835,000
Detroit Edison, Service Area	Fast Breeder	300	100	\$ 9,000,000			600 psi 730°F	1,000,000
Consumers Public Power, Columbus, Nebr.	Sodium Graphite	250	75	\$10,800,000	320	10-11	800 psi 835°F	
Orlando Utilities Comm., Orlando, Fla.	Liquid-Metal		25-40	\$ 8,000,000	512 to 820		1250 psi 900-950°F	250,000 to 400,000
Elk River Rural Co-op, Elk River, Minn.	Boiling Water	58	22 (2)	\$ 2,430,000	281	6.5	615 psi 825 F(2)	225,000

Table 3 Continued

Company & Location	Reactor Type	Reactor Power mw	Generating Capacity mw	Est. Cost Turbine Plant, \$	Est. Total Plant Cost \$/kw cap.	Est. Power Cost mils/kwh	Steam Conditions	Steam Flow lbs/hr
Municipal Power Dept., Piqua, Ohio	Organic-moderated	45.5	12.5	\$1,960,000	424		410 psi 550°F	
Municipal Gas & Elec., Holyoke, Mass.	Gas-Cooled	44	15	\$4,000,000	429		Gas Turbine 1292°F	1,080,000 (nitrogen)
Chugach Elec. Assoc. Anchorage, Alaska	Sodium-Cooled	40	10	\$1,850,000	735		850 psi 850°F	
Wolverine Elec. Co-op, Hersey, Mich.	Homogeneous	31	10(3)	\$1,080,000	356	7.2-9.4	600 psi 825°F (3)	116,000
So. Calif. Edison Santa Susana, Calif.	Sodium-Graphite	20	7.5				600 psi 825°F	
Pacific Gas & Elec., San Francisco Area	Boiling Water		5				1000 psi 545°F	82,000 to turbine

(1) Includes 96-mw oil-fired superheating from 450 to 1000 °F.

(2) Includes 5-mw separately-fired superheating from 450 to 825°F.

(3) Separately-fired superheating from 450 to 825°F.

* Data taken from "A-Power Promises to Compete", Electrical World, McGraw-Hill.

The effect of adding the subcooled water at the reactor inlet is to delay the initiation of boiling in the cooling channels of the reactor. At full load the boiling starts above the mid-point of the channel. The percentage of voids is maintained constant by varying the degree of subcooling, that is, at low loads the water is allowed to boil lower down in the channel. The amount of subcooling is controlled by turbine load through having the speed governing valves located on the secondary steam admission, so that as load increases the pressure drop in the secondary boiler transfers more heat, accomplishing the requisite depression in temperature of the return water to the primary boiler.

The turbine-generator set is a tandem unit with three turbine elements connected to a single generator of 250,000 kva capacity. Flow is away from the generator in the high-pressure element, then through a crossunder to an intermediate-pressure element, flowing toward the generator, then through a crossover connection to a double-flow low-pressure section into a common exhaust and condenser. Moisture separators will be used in the crossover and at each extraction point. The load control governor is on the secondary admission at the intermediate-pressure turbine. The primary admission has an initial pressure governor which controls reactor pressure, with by-pass valves for directing the flow to the condenser in event of turbine trip-out.

Turbine shaft seals are provided for operation on turning gear with the casings filled with water in a decontamination procedure prior to maintenance. The turbine design attempted to minimize crevices for buildup of radioactive particles.

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Fig. 25. General Electric Company's "dual cycle", to be used in Commonwealth Edison plant near Chicago.

Condensate from the main condenser hotwell is pumped first through the gland exhaust condenser and the air removal coolers; the stream then divides, with one part through a demineralizer and feedpump through six stages of feedwater heating and back to the reactor, and the other through another feedpump and through six stages of feedwater heating back to the secondary boiler, as shown in Fig. 25. The feedwater heaters have divided passages to take both flows. The boiler feedpumps are installed in triplicate. A desuperheater is installed in the by-pass from the primary turbine to protect the low-pressure turbine exhaust equipment from the high temperature steam, with an emergency source of water both for this cooling and condensing probably being provided. The demineralizer in the feed is hoped to prevent products of corrosion and erosion from entering the reactor, a development program being under way on high flow rate demineralizers for this service.

The thermal efficiency of this plant suffers from the downgrading of energy due to the secondary boiler heat transfer process. In this case it is considered justified in order to achieve stability at various loads. A base-load plant would minimize the voids problem, and could perhaps be operated at much higher efficiencies.

Army Package Reactor Power Cycle

This reactor power cycle perhaps typifies the small, special system designed to supply power and heat in areas remote from conventional fuel sources (such as the arctic region). It illustrates a feature in which the atomic plant may excel, that of economically supplying large amounts of low temperature steam or water for building space heating or process heat. The following description is based primarily on the feasibility report*

* Boch, A. L., and Livingston, R. S., "A Package Power Reactor for Remote Locations," Paper No. 55-SA-71, Am. Soc. Mech. Engrs.

Fig. 26. Proposed steam cycle for U.S. Army "Package Reactor". Electrical capacity is about 1500-kw. (This cycle has been modified for the Fort Belvoir installation).

rather than on data on the prototype now under construction by the Army at Fort Belvoir, Virginia, but the essential features and philosophies are the same. The flow diagram is shown in Fig. 26.

The reactor is designed around the power requirements, about 1500-kw, and the space heating requirements, about 3500-kw. The pressurized water type reactor supplies heat to a heat exchanger which generates saturated steam at 200 psia. It might be expected that the turbine would be the automatic extraction type with bled steam at about 50 psig used for space heating requirements, but the difference in size and weight between the straight type and the automatic extraction type led to preference of the former, even though there was a sacrifice to cycle efficiency. The heating steam is taken straight from the boiler. The thermal efficiency of the cycle for electric generation is about 15% and including the space heating, is about 45%. One stage of feedwater heating is used, mainly to provide deaeration. An evaporator is included for make-up. Because the conceptual design was for an arctic power station, one requirement was that all equipment must be of size and weight suitable for flying it to the site. The bulky concrete shield, using local aggregate, would also serve as the turbine-generator foundation. Air-cooled condensers presented the problem of freeze-up in the -60°F design conditions, so an antifreeze would be circulated through a shell and tube condenser and through air-cooled coils. The heat transfer was less than adequate in the summer so it was planned that a water source could replace the antifreeze, or the outside coils might be operated wet. All pumps, etc., were installed in duplicate.

Power Cycle for Thermal Breeder Reactor Conceptual Design *

Studies of a 300-kw electrical capacity power plant using three thermal breeder type reactors having 5-ft diameter core and 10-ft diameter blanket pressure vessels, were made by ORNL personnel. A saturated steam cycle was selected and the reactor-turbine plant system optimized using typical power station equipment costs and estimated costs for reactor components and fuel inventories. As could be expected from other studies of steam cycle optimizations, (See Section X), analysis of the turbine plant indicated that thermal efficiency improved as throttle pressure was increased. Because the cost of the three feedwater heaters and piping increased with pressure, a slight upturn to the turbine plant investment cost curve was indicated at about 530°F, as shown in Fig. 27. The core and blanket system investment costs rose rather sharply with temperature, primarily because of decreased temperature differentials in the heat exchangers, so that the total incremental power cost including both the reactor and turbine plants, passed through a minimum at about 400 to 450°F throttle temperature, as may be seen on Fig. 27. Although one must consider the assumptions made in this study, the fact that an optimum temperature of relatively low magnitude was indicated has interesting connotations.

The steam cycle decided upon employs three 100,000-kw turbine-generators, tandem-compounded, double-flow, hydrogen-cooled, units with 335 psia dry and saturated steam to throttle, with three points of extraction for regenerative feedwater heating, one moisture separation at about 26 psia, and exhaust at 1-1/2" Hg abs.

The flow diagram for the system is shown in Fig. 28. The moisture

* Taylor, W. F., "TBR Plant Turbogenerator System Study", Oak Ridge National Laboratory Report No. 56-7-127, June 1956.

Fig. 27. Total turbine plant costs and incremental total power costs (including reactor investment) of proposed Thermal Breeder Reactor (see Fig. 3.28) power station, for various saturated steam throttle conditions. Temperatures on the curves are inlet and outlet of the reactor core.

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Fig. 28. Steam Cycle for Proposed Thermal Breeder Reactor Power Station. Three turbine-generator units of 100,000-kw capacity are proposed.

separators selected were the centrifugal type, estimated to be about 8-ft in diameter and 17-ft long with an inlet connection 40" diameter. The cost was very roughly estimated at \$200,000 (or \$2 per kw of capacity). Other estimated costs are as shown in Table 4.

Separately-Fired Superheater Steam Cycle

Conventional fuel-fired furnaces may be used to superheat the steam from a reactor system in order to use modern steam turbine-generator equipment. The Consolidated Edison plant at Indian Point, New York, rated at 236-mw, is of this type. This arrangement has much the same problems as those of the supercritical cycle in that the oil-fired superheater transfers heat at only a relatively high temperature with little opportunity to cool the exit combustion gases to achieve good efficiency for the combustion unit. In this case, the reactor is supplying the low-availability (and less expensive) type of heat. In addition to allowing use of conventional turbines, the cycle has the advantage of good flexibility for handling load fluctuations. Should the reactor portion of the plant not perform satisfactorily, or become obsolete, the investment in the superheating equipment is still largely intact. Arguments for the cycle are that if one must build a prototype reactor power plant to contribute to development of the program, then an oil-fired superheater may be considered. The cost of power generated by the combination type plant will be greater than for a conventional oil-fired steam plant. The steam flow diagram for this plant is shown in Fig. 29.

Fig. 29. Steam cycle for Consolidated Edison power plant having separately-fired superheater supplying 96 mw of heat

Table 4

ESTIMATED COSTS FOR TBR TURBINE PLANT

	<u>Totals</u>	<u>\$/kw*</u>
Land	\$150,000	\$ 1.69
Structures		
Turbine plant bldg.	\$1,159,400	
Screenhouse bldg.	398,000	
Land improvements	185,600	
Railways	106,000	
	1,850,000	20.99
Boiler and Accessories		
Boiler feed system	183,600	
Water purification	34,700	
Cooling Water System	25,500	
Station piping	545,700	
	789,500	8.92
Turbine-Generator and Accessories		
Turbine-generator	3,100,000	
Erection	150,000	
Foundations	85,500	
Condenser and Aux.	910,000	
Miscellaneous	532,000	
	4,777,000	53.98
Electrical Equipment	816,000	9.22
Misc. Power Plant Equipment	210,000	2.37
Substation Equipment	800,000	0.04
Total for Turbine Plant	\$9,392,700	\$106.12

* Includes 13% additional for indirect costs and overhead.

VIII. PERFORMANCE TRENDS FOR STEAM PLANTS

Since the nuclear powered steam generating station may have difficulty competing with fossil fuel-fired plants until these fuels become more expensive, the designer of the nuclear plant may be more concerned with future prospects for heat power cycles than those based on the present state of the art.

The physical properties of the materials of construction have consistently fixed the limitations of pressure and temperature for the vapor cycle. There has been continued progress in development of new high strength steels for elevated temperatures, as shown in Fig. 30*. If the data plotted in this graph can be said to establish a trend which can be extrapolated, temperatures as high as 1600°F will be realizable within a generation. For best utilization of heat at this temperature, pressures as high as 10,000 psi must also be considered**. This high pressure would require several reheats to control turbine moisture and the stages of feedwater heating would probably reach fifteen or more. Both the complexity of the plant and the high cost of materials to withstand these conditions may be the chief deterrents to the construction of such a plant. The 1150°F temperature in use today requires highly expensive forged and bored stainless steel for the piping, and it appears that further improvements will come with ever-increasing difficulty. Binary type cycles to reduce the pressure accompanying the high temperatures may help to reduce the materials problem, and use of straight-through boilers to eliminate the highly expensive drums may give impetus.

* Maloney, J. D., "Estimated Trend of Thermal Efficiency and Real Cost of Steam Power Stations for the Next Fifty Years", ORNL 54-3-67, Mar., 1954.

** Knowlton, P. H., and Hartwell, R. W., "Possibilities of the Regenerative Steam Cycle at Temperatures up to 1600°F," Combustion, Jan., 1950.

Fig. 30. Progress of turbine throttle temperatures and indication of possible attainment if trend can be extrapolated. (From Maloney, op. cit.)

Fig. 31. Comparative costs of indoor and outdoor steam power stations. (From Electrical World Ninth Steam Station Cost Survey).

It is not only the increased temperatures and pressures that have caused the thermal efficiency of plants to improve over the past years, but better techniques and designs have also been important. The trends in thermal efficiency improvement are shown in Table 5, which also gives the percent of perfection, or percent of attainment of the Carnot efficiency. As this ideal efficiency is more closely approached, improvements become more difficult and losses reach a practically irreducible minimum. Estimates of 80% degree of perfection have been advanced as the maximum probably attainable.

The fact that construction costs have been decreasing (based on constant dollar value) per kilowatt of output is attributable in large measure to use of outdoor installations, to larger turbine-generator units, and to larger central stations. In recent years central steam stations of over 1,000,000-kw installed capacity, while not common, are accepted as posing no unique design problems*. General Electric announced in June, 1956**, the construction of 450,000-kw turbine-generator units, which will be cross-compounded with the high pressure elements operating at 3600 rpm and the low pressure at 1800 rpm. The stator (armature) of the high-speed generators will be cooled by oil circulated through hollow high-voltage windings and the rotor (field) will be cooled by hydrogen. The steam cycle for these turbines will have double reheat, have a maximum pressure of 3500 psia and temperature of 1050°F, and a claimed overall thermal efficiency of 41%.

* McClure, J. B., and Mellor, A. G., "Electric-Power Generation, Past Present and Future", Mechanical Engineering, June, 1956, p. 521.

** "450,000-kw Steam Turbine-Generators", Mechanical Engineering, June, 1956, p. 545.

Table 5

TREND OF PLANT EFFICIENCIES AND COSTS*

Survey Year	Thermal Efficiency		Maximum Throttle Conditions		Carnot Efficiency (1) %	Perfection (2) %	1939 Real Cost (3) \$/Kw
	No. of Plants	Avg. %	Max. %	Pressure psi	Temperature °F		
1927	15	15.4	18.5	600	750	54.4	119.6
1934	13	23.7	30.5	1400	825	57.0	136.2
1936	19	23.2	30.5	1400	900	59.4	125.4
1938	52	26.2	31.7	1400	950	60.8	101.1
1947	106	23.7	32.2	2335	1000	62.2	85.0
1950	47	28.1	34.3	2335	1050	63.4	81.2
1952	41	29.6	36.7	2335	1100	64.6	69.2
1956 ⁽⁴⁾	1	40.0	40.0	4500	1150	65.5	60.1
2000	(Extrapolated)		52.5			77.5	68.0

Value of Dollar

- (1) Based on maximum throttle temperature and 92°F exhaust
 (2) Percent max. thermal efficiency is of Carnot efficiency
 (3) Adjusted for value of dollar and based on rated capacity
 (4) Nearing completion

1927	1.15	1947	0.84
1934	1.24	1949	0.59
1936	1.14	1950	0.46
1938	1.00	1952	0.43

* Maloney, op. cit.

IX. POWER CYCLE COSTS AND TRENDS

Caution must be exercised in drawing conclusions from averages of power station costs, particularly when the averages encompass a wide variety of plant types, fuels, locations and plant load factors. One cause of wide deviations in reported costs is the different philosophies of the utilities in regard to annual charges against investment. Whereas much can be gained from examination of the costs of conventionally-fired plants, cost-wise the atomic power plant is not simply a conventional plant with a nuclear process substituted for the combustion process, but substantially a new situation where established precepts require re-examination. At any event, it is too early to discuss nuclear plant costs in any but general terms, since operating and maintenance cost data are yet to be substantiated.

Conventional Steam Plants

An average of 54 of the most modern steam plants in the U.S.* shows that the 1954 cost of producing power was between 6 and 7 mills per net kwh at the bus bar. About 60% of the plants had costs less than 7 mills, about 16% had less than 4 mills, and about 24% had costs greater than 8 mills. The highest cost was 12.5 mills and the lowest 3 mills. The great spread is due to the methods of charging off plant investment costs, the plant load factors, and site conditions peculiar to a given plant.

A typical breakdown of the national average cost of 6.3 mills/kwh is:

* "Ninth Steam Station Cost Survey", Electrical World, Oct. 17, 1956, McGraw-Hill Pub. Co.

Table 6

Power Generating Costs Large Steam Stations*

	<u>Mills/kwh</u>	<u>Percent</u>
Fuel	2.47	39.5
Wages	0.30	4.8
Water & Supplies	0.06	1.0
Maintenance	0.15	2.4
Fixed Charges	3.28	52.4
	<u>6.26</u>	<u>100.0</u>

It must again be cautioned that the fixed charges shown in the above table are subject to close examination, since a spread of values were reported (from 0.51 mills/kwh to 5.97 mills/kwh) for the fixed charges. The investment charges vary from 3% for public power stations to 17% for private utilities, with about one-half the plants using values between 13 and 14%. The plant load factor also has a great influence on the fixed charges. The 1954 average load factor was 71.6%, but it must be remembered that the newer, more efficient plants tend to carry the base loads and hence show low power costs as a result of the lower fixed charges.

Average investment for 54 modern stations* showed that 50% cost less than \$129/kw and 30% cost more than \$140. Extremes were as low as about \$80/kw and as high as \$220/kw. Outdoor plants are about \$75/kw cheaper than indoor types for plants less than 65-mw capacity and about \$17/kw cheaper

* Ninth Steam Station Cost Survey, op. cit.

for plants over \$150-mw size, as shown in Fig. 31. Maintenance on an outdoor plant averages only slightly higher than for indoor types.

The investment cost may be broken down as shown in Table 7, but there may be wide variations between plants.

Table 7

Investment Costs in Large Steam Stations**

	FPC Acct. No.	Percent of Total Investment	
		Gas-Oil Fired	Coal Fired
Land	310	less than 1%	less than 1%
Structures & Improvements	311	17.3	24.2
Steam Generating Equipment	312	37.2	40.0
Turbine-Generator	314	35.9	27.4
Electric Equipment	315	8.0	7.2
Miscellaneous	316	1.5	1.2
		100.0	100.0

It may be noted in Table 7 that the cost of steam generation equipment is the largest part of the investment. It is also the most expensive part of the plant to maintain, the average maintenance cost being about 0.07 mills/net kwh as compared to about 0.05 mills/net kwh maintenance costs for turbine and auxiliary equipment.

Figure 32 shows that as station heat rates have been lowered through better efficiencies, the improved performance plants have also been more expensive, based on existing dollar values. Figure 32 also shows that as the cost of heat (fuel) increases, justified investment cost to conserve the heat also increases.

** Based on data, "Ninth Steam Station Cost Survey", op. cit.

Fig. 32. Trend of station cost and heat rate as function of the prevailing throttle pressure. (From Electrical World, Ninth Steam Station Cost Survey).

The twenty-five year trend has been toward decreasing investment costs as plant sizes and efficiencies have increased, as shown in Table 8, if costs are adjusted for the value of the dollar.

Table 8

Trend in Large Steam Station Investment Cost*

Year	Average Investment Cost, \$/kw	Value of** Dollar	Real Cost \$/kw
1927	\$ 104.00	\$ 1.15	119.6
1934	109.8	1.24	136.2
1936	110.0	1.14	125.4
1938	101.1	1.00	101.1
1947	101.2	0.84	85.0
1949	137.6	0.59	81.2
1950	150.4	0.46	69.2
1952	139.8	0.43	60.1

Nuclear Plants

Whereas public utilities have been striving for higher and higher temperatures in their steam plants, the nuclear designer is still largely influenced by the reactor design limitations. If steam plant investment cost data are modified to exclude the steam generating and fuel handling equipment, it is apparent that large modern plants are approaching a minimum total investment cost per net kilowatt installed capacity, see Fig. 33. In other words, the future large atomic plants cannot expect much improvement in costs of the power cycle equipment from sheer size alone.

* Maloney, J. D., "Estimated Trend of Thermal Efficiency and Real Cost of Steam Power Stations for the Next Fifty Years", Oak Ridge National Laboratory Report No. 54-3-67, March 11, 1954.

** Based on Engineering News Record construction costs with 1939 = \$1.00.

The cost of power from the turbine-generator may be estimated if a cost of steam is known, regardless of whether its source is from nuclear or combustion processes. This method may be used to estimate how much capital can be economically invested in a reactor plant. The cost of steam in public utility plants varies considerably with plant location. In the southwestern U. S., natural-gas-fired plants produce steam costing as low as \$0.15 per million Btu, and in the northeastern parts as much as \$0.50 per million Btu.

Since the nuclear plant designer, at least for the present, is often interested in saturated steam, studies have been made to adjust cost of steam turbine-generators to saturated steam conditions. The cost of electricity as a function of the steam cost was determined in one such study* by analyses of several existing steam plants, with appropriate adjustments for saturated conditions. These values, with associated steam conditions, are shown in Figure 33. The zero steam cost curve indicates the cost of electricity due to investment, maintenance, etc., even though steam was available at no cost. Now, if the steam source is a reactor costing \$100 per kw or 2.14 mills/kwh based on 15% investment cost and 80% load factor, the total cost of electricity would be about 6.2 mills/kwh with 400°F throttle conditions. It may be noted from Fig. 33 that a conventional plant operated at about 1000°F could generate electricity at about 6.2 mills if steam costs were about \$0.40/mBtu, a value typical of high-fuel cost regions. If the same \$100/kw reactor were operated at 600°F, with an estimated total generating cost of 5.5 mills/kwh, a conventional 1000°F plant could compete if steam cost \$0.30/mBtu.

* Maloney, J. D., "Cost Estimates for Seven 200-MW Turbine Plants for Operation with Nuclear Reactors at Various Steam Conditions", Oak Ridge National Laboratory Report No. ORNL-1386, Dec., 1952.

Fig. 33. Calculated production cost of electricity as function of steam cost for various throttle temperatures. Numbers on curves are steam costs in dollars per million Btu's. Based on 80% plant factor and 200-mw size plant. (From Maloney, *op. cit.*).

Fig. 34. Investment cost of various accounts in 200-mw turbine plant for various saturated steam throttle temperatures. (From Maloney, *op. cit.*)

Since investment costs for the reactor portion of the plant may be as much as \$300/kw in the initial stages of development, and fuel costs for the reactor cannot be considered negligible when fuel processing is included, the above study indicates that atomic power may be competitive with combustion-type power plants only after considerable cost reductions are made, and then perhaps only in high fossil fuel-cost regions.

This same study* estimated cost of turbine-generators for saturated steam cycles from manufacturer's data on low-pressure elements of large tandem, double-compounded, 1800 rpm, units. These estimating values, together with typical estimating costs for other equipment in the saturated cycle, land, structures, improvements, electrical gear, etc., are shown in Fig. 34.

The special safety precautions required for a nuclear plant are estimated to add about \$10/installed kw to plant costs**.

* Maloney, J. D., op. cit.

** Lane, James A., "Reducing Nuclear Power Costs", Nucleonics, Jan., 1955, McGraw-Hill.

X. OPTIMUM CYCLES FOR NUCLEAR PLANTS

The trend of conventionally-fired large steam power plants has been toward larger sizes and higher temperatures and pressures. The optimum cycle for a given reactor system tends to be the same direction, but if the designer is limited to saturated steam, then the picture is changed because of the rapidly rising cost of equipment with higher pressures which accompany only relatively small gains in temperature. If the reactor is of the breeder type, where steam costs might be considered negligible, the picture is again altered.

Several studies have been made on optimization of saturated cycles, some based on adjustment of cost values of an actual power station and some on mathematical treatments based on national averages for cost of equipment and design conditions. The broad assumptions requisite for such studies probably limit their usefulness, but trends may be indicated.

The study previously mentioned* considers the influence of throttle pressure on costs, and is predicated on data from actual plants. Considering the cost of prime mover equipment for a saturated steam cycle, as shown in Fig. 34, it is noted that there is no indication of an optimum throttle pressure, with the possible exception of the feedwater system and piping. This particular account is influenced by the fact that as pressure is increased, costs are increased, but size is decreased due to greater density of the steam. Figure 34 indicates that the total production cost continues to decrease as the steam pressure to throttle is increased, although at a dwindling rate. The study concluded that a trend toward higher pressures was

* Maloney, J. D., "Cost Estimates for Seven 200-Mw Turbine Plants for Operation with Nuclear Reactors at Various Steam Conditions", Oak Ridge National Laboratory, ORNL-1387, Dec. 1952.

desirable for saturated cycles.

Another study* defines incremental cost of heat from a reactor, C_Q , as that cost required to produce an additional Btu, including the additional incremental operating and fixed charges associated with it. Assuming the usual 7000 hr/yr full-load operation and 15% investment cost for a 150-Mw plant:

$$\frac{C_Q \times 150,000 \times 7000}{0.15} \times \frac{\partial \text{HR}}{\partial I} = -1.0$$

where C_Q is the cost of reactor heat, \$/Btu, and ∂I is the increment of turbogenerator plant cost in dollars, necessary to produce ∂HR , the increment of plant heat rate, expressed as Btu/kwh. This equation states that the marginal, or limiting, decrease in annula operating cost is capitalized at the amount of the corresponding increment of plant investment.

Since the factors influencing the investment costs and heat rates do not lend themselves to simple mathematical relationships, the above expression was treated graphically. Further, it is necessary to make broad assumptions to relieve the problem of as many variables as possible.

Some of these assumptions were:

- a. Turbine costs are directly proportional to exhaust annuli areas.
 - b. Condenser costs are directly proportional to heat transfer surface, and assumed to be \$21.78/ft², including overhead, etc.
 - c. The turbine has uniform stage efficiencies of 82%, an exhaust moisture content of 12% and a condensing temperature of 80°F.
- Leakage, windage, mechanical and electrical losses are therefore neglected.

* "Reactor Steam Power Plants - Economics", AECD-3659, Spet. 1953, Nuclear Engineering Project, Mass. Inst. Tech., Cambridge, Mass.

Fig. 35. Optimum plant investment for various steam costs as function of the throttle temperature. Numbers on curves are steam costs, dollars per million Btu's. (From "Reactor Steam Power Plants - Economics", op. cit.).

Fig. 36. Optimum calculated plant heat rates for various steam costs as function of throttle temperature. Numbers on curves are steam costs, dollars per million Btu's. (From "Reactor Steam Power Plants - Economics", op. cit.)

- d. Moisture separators are assumed 90% effective in removing water and have a Δp of 5% of initial pressure.
- e. Foundations, building, etc., are assumed at \$12,400 ft², including overhead and contingencies.
- f. Feedwater heaters are estimated at \$69.70 per 1,000 lbs/hr, including overhead.

The analyses first balanced turbine exhaust annuli against condensing surface with exhaust enthalpy as the common denominator. The optimum number of feedwater heaters was determined by Salisbury's* methods.

Results of the study are shown in Fig. 35 and Fig. 36, for the case of the regenerative saturated cycle employing moisture separation. The conclusions to be drawn correlate those of the Maloney study mentioned above, that plant investment costs must be kept low if nuclear plants operating on saturated steam are to generate competitively with conventional fuel fired plants. Analyses of other cycles, including superheat, were included in the reference study.**

* Salisbury, J. K., Steam Turbines and Their Cycles, John Wiley and Sons, 1950.

** "Reactor Steam Power Plants - Economics", op. cit.

SECTION XI. DEFINITIONS

Capacity

The capacity factor is the ratio of the average load on a generator for a specified interval of time to the nameplate rating of the unit.

Compound Turbines

Because the specific volume of low pressure steam becomes relatively large, it is usually necessary to divide the flow at some point in its expansion through a turbine into two or more streams for passage through low-pressure turbines. If the low pressure turbines are on the same shaft as the high-pressure turbine it is said to be tandem compounded. If the low-pressure turbines have separate shafts with their own connected generators, it is cross compounded.

Condensing Turbine

A condensing turbine is one that operates with an exhaust pressure less than atmospheric. A non-condensing turbine is one that operates with greater than atmospheric pressure exhaust, even though it may discharge into a steam condenser.

Demand

The demand is the average load at the station terminals over a specified interval.

Maximum demand is the greatest load which occurred during a specified interval.

Double Flow Turbines

Double flow steam turbines have the entering steam flow split and flowing through the casing in opposite directions. This arrangement reduces the thrust on the shaft.

Efficiency

Overall plant efficiency is the ratio of the station net power output for a given interval of time to the total heat contained in the fuel fired during that period.

Thermal efficiency of a plant is the ratio of net power output over a given interval to the heat delivered to the working fluid over the same period.

Turbine efficiency is the ratio of the actual net work done by a unit weight of the working fluid to the work that would have been done by the fluid had the process been isentropic. All heat and friction losses and the exhaust loss are included.

Internal efficiency is the same as turbine efficiency except that the exhaust loss is not included.

Stage efficiency is the ratio of work done by a unit weight of the working fluid in passing through a turbine stage to the isentropically available work of expansion. It does not include any heat or frictional losses, nor the leaving energy of the stream.

Elements

Compounded turbines are divided into two or more separate turbines, usually called the high-pressure element and the low-pressure element. An element refers to the complete turbine, its generator and condenser.

Extraction Turbine

When a turbine is arranged with openings in the casing, or extraction nozzles, for bleeding steam from the turbine at various stages in its expansion, it is called an extraction type turbine. Turbine nozzles and blading must be carefully designed for given flow rates so it is important that the amounts extracted not be varied markedly from design conditions if turbine efficiency is not to be impaired. Extraction turbines may be fitted with

a valve outside the casing, or more commonly, an internal grid valve immediately downstream from the extraction point which may be rotated to maintain the pressure constant in the extraction nozzle even though the turbine loading conditions are changed. These types are called single automatic extraction turbines if there is one controlled extraction point, double automatic extraction if two, etc. The single type is far more common.

Governing Valves

Turbine governing valves control the admission of steam to the turbine in response to speed change so that the rotative speed is held constant. On units of any size, multiple lift types are used which are a series of valves which are lifted successively as more steam is needed admitting steam to additional nozzles. The governing valves are most often employed to control steam to the first stage, but may be used to admit steam at successive stages. When an AC turbine-generator set is operating in phase with a large power network, the governor does not technically control the speed, since the generator is held in synchronism by the connected system, but is used to control turbine-generator power output.

Heat Rate

The overall heat rate of a plant is the average power output over a specified interval, expressed in Btu/hr, divided by the overall thermal efficiency of the plant.

Load

The load on a turbine-generator is its power output. Base load is the minimum load over a specified interval of time. Connected load is the sum of the continuous ratings of equipment to the system. Peak load is the maximum power produced over a specified interval, and it may be the instantaneous or average value. Load factor is the ratio of the average load over a period

of time to the peak load occurring in that period.

Mechanical Drive Turbine

Turbines used for direct drive of pumps, fans, etc., are called mechanical-drive units and are usually smaller and less efficient than those used to drive electric generators.

Plant Factor

Plant factor is the ratio of the average load over a specified time to the total of the ratings of all the generators in the plant.

Single Flow Turbines

Single flow steam turbines have the steam flowing through the casing in one direction only, say toward the generator end. See double flow turbines.

Steam Chest

The space immediately ahead of the governing valves on a turbine.

Stop Valve

Turbine stop, or interceptor, valves are between the throttle valve and the steam source to provide positive shutoff of the turbine. They are not flow control valves.

Throttle Valve

A flow control valve, often hand-operated, between the stop valve and the governing valves on a turbine, is used to control turbine speed during starting operations when the generator must be brought into synchronism with other units already on the line. Sometimes throttle valves and stop valves are combined into one valve assembly.

Utilization Factor

Utilization factor is the ratio of the maximum demand of the system to the rated capacity of the system.

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